(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 646 703 B1

(12) # EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention of the grant of the patent:
10.12.1997 Bulletin 1997/50

(51) Int Cl.6: **F02B 75/02**, F02D 13/02

(21) Application number: **94115395.9**

(22) Date of filing: **29.09.1994**

(54) **Power train having supercharged engine**

Antriebsstrang mit aufgeladenem Motor

Arbre de puissance avec moteur suralimenté

(84) Designated Contracting States:
DE FR GB

(30) Priority: **30.09.1993 JP 245057/93**
30.07.1994 JP 197400/94

(43) Date of publication of application:
05.04.1995 Bulletin 1995/14

(73) Proprietor: **Mazda Motor Corporation**
Aki-gun Hiroshima-ken (JP)

(72) Inventors:
- **Goto, Tsuyoshi**
  Hiroshima-shi, Hiroshima-ken (JP)
- **Sugimoto, Hiroyuki**
  Aki-gun, Hiroshima-ken (JP)

(74) Representative: **Heim, Hans-Karl, Dipl.-Ing. et al**
Weber & Heim
Patentanwälte
Irmgardstrasse 3
81479 München (DE)

(56) References cited:
EP-A- 0 269 125
DE-A- 2 544 766
US-A- 5 239 960

- **AUTOMOTIVE ENGINEERING, vol. 101, no. 7, 1 July 1993 pages 76, 79-81, XP 000387353 JY 'Concepts : Japanese "Miller-cycle" engine development accelerates'**

EP 0 646 703 B1

Description

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a power train having a multi-valve double overhead camshaft (DOHC) engine equipped with a mechanical supercharger and a transmission.

2. Description of the Related Art

In order to increase the amount of air to be forced, under pressure, into an engine and thereby to increase engine output torque, engines are typically equipped with superchargers. In addition, there have been known what are called mirror cycle engines in which closing of intake valves is retarded greatly after bottom dead center so as to make effective compression ratio smaller than expansion ratio, thereby reducing the work of compression and reducing pumping loss. In recent years, the idea has been proposed of using this kind of method in engines equipped with superchargers and intercoolers with the aim of controlling knocking and boosting engine output torque. Such a Miller cycle engine is known, for instance, in Japanese Unexamined Utility Model Publication No. 63 - 51121.

In the engine described in the above-mentioned publication, which has double overhead camshafts and is equipped with a supercharger and an intercooler provided in an intake passage, the time at which intake valves close is retarded later than 70° after bottom dead center. With this engine, a reduction in effective compression ratio provides the effect of lowering the temperature at top dead center of compression stroke, so that knocking and temperature rise of exhaust gases are controlled and under these controlled conditions, the supercharging volumetric efficiency is increased and engine output torque is effectively increased.

Engine output is transferred to wheels of the vehicle via a power transfer device, including a transmission and a reduction gear, which is constituent elements of a power train as well as an engine. Power trains installed in conventional passenger cars and the like, even if they includes supercharged engines, have overall gear ratios of about 3.15 for the highest gears i.e. at the smallest gear ratios, of transmissions, while having the maximum speed of no less than 6,500 rpm. for a maximum engine output. In DOHC engines, in particular, because the DOHC mechanisms provide improved valve lifting performance at high speeds, the engines have increased maximum speeds so as to attain horsepower.

The conventional power trains of this kind achieve high engine output and, however, provide poor fuel economy and room for improvement in terms of noise, reliability and the like.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide a power train including an engine equipped with a mechanical supercharger that secures power train output and maintains good driving performance, while providing greatly improved fuel economy through increased engine output torque and lowered engine speed and boosting reliability.

The above object of the present invention is achieved by providing a power train having a power transfer line and an engine, which is desirably equipped with double overhead camshafts, cooperating with a mechanical supercharger and an intercooler disposed in an intake line. The engine is designed and adapted to have a time of closing intake valves no earlier than a crank angle of 65° after bottom dead center and a maximum engine speed no greater than 6,000 rpm for a maximum output. The detailed engine specifications are set so that the maximum engine output accords to a maximum volume of charge of the mechanical supercharger. Further, the power transfer line is designed and adapted to have an overall gear ratio between 2.1 and 2.8 for the highest gear of the transmission coupled to the engine.

If the mechanical supercharger is, preferable in the power train of this invention to be, of an internal compression type, the engine is adapted to have a geometrical compression ratio or mechanical compression ratio (i.e., as determined by mathematical calculation from a fixed volume) of no less than 8.5. It is desirable for the engine to have a displacement of no greater than approximately 2,500 cc and, in addition, to be provided with a plurality of intake ports on each cylinder.

Furthermore, a desirable engine of the power train of this invention is one that has a plurality of intake ports and a plurality of exhaust ports on each cylinder, with the total area of the intake port openings being larger than the total area of the exhaust port openings. In addition, it is effective for the power train of this invention to include a transmission whose gear ratio in the highest gear is less than 1.0.

With the power train of the present invention, knocking is controlled due to a retarded timing of closing the intake valves and a reduction in effective compression ratio, while the engine output torque is increased, in particular effectively at low engine speeds, through supercharging performed by a mechanical supercharger. Furthermore, while the engine output torque is thus increased, the power train is adapted to have an overall reduction ratio appropriately reduced

and a decreased maximum engine speed, thereby achieving driving conditions beneficial to fuel economy and the like while maintaining a desired maximum output. In other words, as will be explained in detail hereinafter in connection with preferred embodiments, when compared with conventional models, the engine operating region that achieves equivalent driving or traveling conditions shifts toward the side of lower speeds and high torque, which is beneficial in terms of fuel economy. Furthermore, due to the use of lower speed engines, beneficial effects are achieved in terms of noise reduction and improved reliability.

In the power train of the present invention, an internal compression type of mechanical superchargers provide adequately increased pressure. In addition, engines having geometrical compression ratios of no less than 8.5 attain favorable expansion ratios and, due to a retardation of closing the intake valves, appropriate effective compression ratio, leading to a more profitable effect in increasing output torque.

When the engine has a displacement of no greater than 2,500 cc, benefits are achieved in terms of fuel economy, while an engine output comparable to an engine having a large displacement can be obtained by a high volume of supercharge. In addition, when the engine has a plurality of intake valves on each cylinder, intake air resistance during a high volume of supercharge is reduced due to the increased area of intake openings compared with engines provided with only one intake port on each cylinder, so that a high volume of supercharge is beneficial in terms of boosting torque. In this case, when the engine has a plurality of exhaust ports, in addition to a plurality of intake ports, on each cylinder whose total area of openings is larger than that of the intake ports, intake air resistance is more effectively reduced.

Furthermore, when the engine used in the power train of the present invention, is of a type having double overhead camshafts, freedom in laying or positioning the intake and exhaust valves is increased, so as to enable the intake ports and intake valves to have shapes and angles favorable to a reduction in the resistance of air flow at the intake port openings, thereby achieving a beneficial effect in terms of reducing intake resistance during a high volume of supercharge.

The power train having a transmission whose highest gear has a gear ratio of less than 1.0 prevents a terminal reduction ratio from being excessively small, even if the transmission coupled to the engine has the highest gear whose overall gear ratio is between 2.1 and 2.8.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other objects and features of the present invention will be clearly understood from the following description with respect to the preferred embodiment thereof when considered in conjunction with the accompanying drawings, in which:

Figure 1 is a schematic plan view of a power plant equipped with a supercharged engine in accordance with a preferred embodiment of the present invention;
Figure 2 is a schematic cross-sectional view of the supercharged engine;
Figure 3 is a timing diagram of opening and closing intake valves;
Figure 4 is a diagram regarding the definition of closing timing of an intake valve;
Figure 5 is an iso-horsepower diagram relative to engine speed, and necessary torque and overall reduction ratio of the power transfer line with a transmission in the highest gear;
Figure 6 is a diagram showing the relationship between fuel efficiency and overall gear ratio of the power transfer line with the transmission in the highest gear;
Figure 7 is a diagram showing the relationship between time needed for acceleration to 100 km/h and overall gear ratio of the transfer line with the transmission in the highest gear;
Figure 8 is a combined diagram of Figures 6 and 7;
Figure 9 is an iso-fuel economy diagram;
Figure 10 is a diagram regarding necessary acceleration time; and
Figure 11 is a diagram showing the relationship between noise and engine speed.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to the drawings in detail, and, in particular, to Figures 1 and 2 schematically showing an engine 1, such as an internal combustion gasoline engine, equipped with a supercharger 20, both of which constitute essential elements of a power train in accordance with a preferred embodiment of the present invention. The engine 1, which has a high geometrical compression ratio or mechanical compression ratio (i.e., as determined by mathematical calculation from a fixed volume) of no less than 8.5, is equipped with a plurality of cylinders 2. In each of the cylinder 2, a combustion chamber 4 is formed above a piston 3 received in a cylinder bore. A plurality of, for instance two in this embodiment, intake ports 5 open into the combustion chamber 4. Similarly, two exhaust ports 6 open into the combustion chamber

4. The total area of openings of the intake ports 5 is designed and adapted to be larger than the total area of openings of the exhaust ports 6. Each of the intake ports 5 and each of the exhaust ports 6 are opened and closed at an appropriate timing by an intake valve 7 and an exhaust valve 8, respectively.

A valve lifter that drives intake valves 7 and exhaust valves 8 has a pair of camshafts 9 and 10 for the intake valves and the exhaust valves, respectively, extending over a cylinder head (not shown). This valve lifter is of such a direct drive double overhead camshaft type that the intake valves 7 and the exhaust valves 8 are directly driven by cams formed integrally with the camshafts 9 and 10. In the center of the combustion chamber 4 a sparkplug 11 is provided.

The engine 1 is provided with an intake passage 12 through which intake air is introduced into the engine 1 and which comprises a common intake route 13 forming an upstream part and an intake manifold 14 forming a downstream part. The intake manifold 14 has discrete intake passages 15 separate for the respective combustion chambers 4. Each of these discrete intake passages 15 is divided at the downstream end into two outlet passages in communication with the respective intake port 5. Each of the exhaust ports 6 is in communication with discrete exhaust passages 26 of an exhaust manifold 25.

The common intake passage 13 of the intake passage 12 is provided in order from its inlet end with an air cleaner 16, an air flow meter 17 for detecting the amount of intake air, and a throttle valve 18 that is operated by means of an accelerator pedal (not shown). Downstream from the throttle valve 18, a mechanical supercharger 20, which is preferably of an internal compression type, and, in the preferred embodiment, of a Lysholm type, is provided in the intake passage 12. This mechanical supercharger 20 is driven by a drive mechanism 21 linked to an engine crankshaft (not shown) via, for instance, a belt or the like. Downstream from the supercharger 20 an air-cooled intercooler 22 is provided in the intake passage 12. Either outlet passage of each discrete intake passage 15 is provided with an injector 24 that injects fuel into the combustion chamber 4. If necessary, it may be preferred to provide in either outlet passage of each discrete intake passage 15 a switch valve 24 that closes in a region of low engine load.

The multi-valve, double overhead camshaft (DOHC) engine 1 equipped with the mechanical supercharger 20 thus structured is linked with a power transfer line so as to form the power train. This power transfer line includes a transmission 31 connected to the output of the engine 1 and a terminal reduction gear 33, such as a differential or the like, connected to the transmission 31 via a drive shaft 32. Drive torque is transferred to wheels 35 of the vehicle via axle shafts 34 operationally connected to the terminal reduction gear 33. This power plant is installed in a passenger car, and accordingly, because tires of the wheels 35, which are of a general size used on passenger cars, have an effective radius of around 0.27 - 0.32 m.

The engine 1 may be of a type having intake ports only one for each cylinder, and, in addition, may cooperate with valve lifters of a type other than a double overhead camshaft DOHC type, for instance a type having a single overhead camshaft. However, as will be described below, the engine 1 is desirable to have a plurality of intake ports for each cylinder and of a double overhead camshaft type for the purpose of reducing intake resistance during supercharging.

In the power train thus structured, a specific relationship is provided among intake valve close (IC), the overall reduction ratio of the power transfer line with the transmission 31 in the highest gear, i.e. at the smallest gear ratio, the maximum speed for a maximum output (or a maximum horsepower) of the engine 1, and the maximum volumetric output of the mechanical supercharger 20 as described below. Hereinafter, the term "overall reduction ratio" shall means and refer to the overall reduction ratio of the power transfer line with the transmission 31 in the highest gear, i.e. at the smallest gear ratio, in other words, the reduction gear ratio that is the combination of the transmission gear ratios of the transmission 31 in the highest gear and the reduction ratio of the terminal reduction gear 33.

Referring to Figure 3 showing schematically intake valve opening and closing or valve lift, the intake valve 7 opens near before the piston reaches top dead center (TDC) on the power stroke and closes after bottom dead center (BDC) on the exhaust stroke. Specifically, the intake valve 7 closes about more than 65 degrees, and preferably less than 100 degrees, after bottom dead center (BDC), which is larger compared to regular engines.

The term "intake close (IC) time" used herein shall mean and refer to the time the intake valve can be considered to be substantially closed, for instance, the time the valve has closed as far as an extremely small lift corresponding to the height of a cam ramp or a cam lobe base of a cam of the valve lifter. In the present embodiment, the intake close (IC) time is taken at a valve lift of 0.4 mm as shown in Figure 4.

The reason the intake valve close (IC) time is set to larger than 65 degrees of crank angle after bottom dead center (BDC) is to adequately produce effects such as controlling knocking and the like due to lowering the temperature at top dead center (TDC) on the compression stroke as will be described later, and the reason for limiting it less than 100 degrees of crank angle after bottom dead center (BDC) is because of concerns that intake close times larger than the limit angle provides a drop in the temperature at top dead center (TDC) on the compression stroke below the limit of ignition, causing difficulty in engine ignition.

When the power transfer line has an overall gear ratio between 2.1 and 2.8, and includes, for instance, a four speed automatic transmission with a gear ratio of about 0.7 in a fourth gear as the transmission 31, the terminal reduction ratio of the terminal reduction gear 33 is set between about 3.0 and 4.0. In addition, the engine 1 has a maximum speed for the maximum output set no greater than 6,000 rpm, and the mechanical supercharger 20 has a maximum

volume of charge so that the engine 1 puts out the maximum output. Comparing these settings with a conventional models of this type of power trains equipped with, for instance, double overhead camshaft (DOHC) engines, which has a total reduction ratio of about 3.15 and a maximum engine speed for the maximum engine output of around 6,500 to 7,500 rpm, the power train of the present invention has a smaller overall reduction ratio and a lower maximum engine speed for the maximum engine output.

With the power train of the present invention, knocking is controlled while engine output torque is boosted thanks to the intake valve closing (IC) time more than 65 degrees after bottom dead center (BDC) and the provision of both mechanical supercharger 20 and intercooler 22. In other words, while intake air is forced by the supercharger 20 into the combustion chamber 4 after being cooled through the intercooler 22, and the effective compression ratio is smaller than the expansion ratio because of the retarded intake valve closing (IC) time, causing a drop in the temperature at top dead center (TDC) on the compression stroke. In particular, when the geometrical or mechanical compression ratio of the engine 1 is above 8.5, which is higher than those of general engines equipped with superchargers which are approximately 7.5 to 8.5, the thermal efficiency is boosted while a gain in expansion ratio is attained, and besides, the effective compression ratio drops appropriately due to the retardation of the intake valve closing (IC) time to no less than 65 degrees of crank angle after bottom dead center (BDC).

Through these effects, knocking during a high volume of supercharge and a rise in the temperature of exhaust gas is controlled and, in addition, an increase in engine output torque is provided because of supercharge. Furthermore, when the supercharger 20 is of a mechanical type and, in particular, of a Lysholm internal compression type, supercharging pressure is sufficiently boosted even at relatively low engine speeds. In addition, a reduction in the intake air resistance during supercharging is provided when the engine is of a multiple intake valve type and, furthermore, is enhanced when it is of a double overhead camshaft type. That is, when the engine 1 has a plurality of intake ports 5 on each cylinder, the total area of intake openings is large, so as to provide a reduction in the intake air resistance. When the engine 1 has a plurality of intake ports 5 and a plurality of exhaust ports 6 besides on each cylinder and the total area of intake port openings is larger than that of exhaust port openings, the reduction in the intake air resistance is more sufficiently enhanced. Furthermore, when the engine 1 is of a type having double overhead camshafts, freedom in laying or positioning the intake and exhaust valves is enhanced, so as to enable the intake ports and intake valves to have shapes and angles favorable to a reduction in the resistance of intake air flow at the intake port openings, thereby achieving a beneficial effect in terms of reducing intake air resistance during a high volume of supercharge. This leads to enhanced supercharging performance and a great increase in engine output torque.

In this instance, the power train according to the present invention, which has an appropriately reduced overall reduction ratio and a lowered maximum engine speed for a maximum engine output while providing an increased output torque at low engine speeds, sustains favorable driving conditions while providing an improvement of fuel efficiency or economy, driving reliability, and quietness which will be described hereafter.

Referring to Figure 5, there are shown lines B1, B2 and B3 representing necessary engine torque needed to obtain a specified horsepower for various overall reduction ratios and an iso-horsepower line A corresponding to a required maximum horsepowers. The required maximum horsepower is based on a maximum horsepower of a conventional power train equipped with an double overhead camshaft (DOHC) engine with a standard supercharger, and, more specifically, it is based on the case wherein the power train includes a supercharged, double overhead camshaft (DOHC) engine whose intake valve closing (IC) time is not retarded and whose maximum speed for the maximum horsepower is increased to a speed of approximately 7,500 rpm which is a permissible speed from the standpoint of reliability and the like, and has an overall reduction ratio of around 3.15. The necessary torque to achieve such a maximum horsepower around this level is represented by line B1 for an overall reduction ratio of 3.15, by line B2 for an overall reduction ratio of 2.8, and by line B3 for an overall reduction ratio of 2.1. The range of necessary torque is defined between lines B2 and B3 for the range of overall reduction ratios (2.1 - 2.8) of the power train according to the present invention. Line B1 shows the necessary torque for an overall reduction ratio of the conventional power train.

It is apparent in Figure 5 that, as the overall reduction ratio decreases, the necessary torque is increased and the maximum engine speed for a required maximum horsepower decreases and that when the overall reduction ratio is no greater than 2.8, the maximum engine speed is no greater than 6,000 rpm. Although, in regular engines with superchargers in which the intake valve closing (IC) time is not retarded, an increase in output torque is limited because of knocking and, consequently, it is difficult to increase the engine output torque to the necessary torque for the overall reduction ratio is no greater than 2.8. However, in the power train of the present invention which includes the engine 1 with the mechanical supercharger 20 having a retarded intake valve closing (IC) time, the engine output torque is sufficiently increased so that the necessary torque is attained even for overall reduction ratios less than 2.8.

In this instance, the double overhead camshafts are used in conventional models of high speed engines having the maximum engine speeds for a required maximum horsepower greater than 6,500 rpm in order to reduce inertia of related valve lifters. While in contrast, it is used in the power train of the present invention in order to reduce intake air resistance, so that although the engine of the power train is of a double overhead camshaft (DOHC) type, it has the maximum engine speed for required maximum horsepower less than 6,000 rpm. The engine horsepower follows the

volume of charge from the supercharger, and the maximum volume of charge from the supercharger 20 is adapted so as to provide the required maximum horsepower of the engine 1.

When the necessary torque is produced corresponding to the overall reduction ratios so as to attain the equivalent required maximum horsepower in the manner described above, the relationship between overall reduction ratio and fuel efficiency economy is given as shown in Figure 6, and the relationship between overall reduction ratio and time needed to accelerate the vehicle from 0 to 100 km/h is given as shown in Figure 7. The acceleration time is a barometer for the feeling of driving the vehicle, with better driving feeling indicated by shorter times.

Improvements in fuel economy and starting acceleration performance brought by the power train of the present invention will be described hereafter.

As shown in Figure 6, the fuel economy is variable. That is, the fuel economy is poorer on the side of large overall reduction ratios, improves to a certain point as the overall reduction ratio decreases, and tends to worsen as the overall reduction ratio becomes smaller. The best fuel economy is attained at an overall reduction ratio of around 2.1. The reason for this is explained with reference to Figure 9 which shows iso-fuel economy lines with respect to driving conditions. As indicated by the iso-fuel economy lines, the fuel economy is best in a region of relatively high engine speeds and low engine load. Furthermore, although if the engine is used in the conventional power trains are frequently driven at higher speeds out of the best or peak fuel economy region of speeds, it reduces the frequently used engine speed with a decline in the overall reduction ratio and approaches speeds of the best fuel economy region. Describing more specifically, as shown in Figure 5, with the power train of the present invention which provides the overall reduction ratio smaller than that of the conventional models indicated by line B1 and attains through supercharging a torque required to be higher accompanying the decline in overall reduction ratio, driving conditions needed to provide an equivalent driving performance (which shall means a driving performance under the same horsepower and vehicle acceleration) shifts toward the side of lower speeds and higher torque as compared to the conventional power trains. Accordingly, a region of normal driving in which driving is frequently made shifts toward the side of lower speeds and higher torque, reaching the best fuel economy region shown in Figure 9 and consequently, fuel economy is improved. However, if the overall gear ratio is made small in excess, fuel economy worsens because of driving at lower engine speeds out of the peak fuel economy region.

In addition, as shown in Figure 7, the required time for acceleration is longer on the side of lower overall reduction ratios, becomes shorter to a certain point as the overall reduction ratio is increased, and tends to become longer as the overall reduction ratio is further increased. It becomes shortest at an overall reduction ratio of around 2.8. The causes are revealed by reviewing Figure 10, which shows how the increasing speed of vehicle caused by acceleration changes in accordance with changes in terminal reduction ratio to which the overall reduction ratio relates and shows schematically changes in vehicle speed from 0 to 100 km/h during maximum acceleration for terminal reduction ratios of the same terminal reduction gear, namely FGR1, FGR2 and FGR3 (FGR1 < FGR2 < FGR3). As apparent from Figure 10, when the terminal reduction ratio changes from FGR1 to FGR2, when the number of gear shifts is the same, as the terminal reduction ratio becomes larger, the rising of vehicle speed is faster, making the acceleration time shorter. When the terminal reduction ratio further changes to FGR3, the number of gear shifts increases, the acceleration time is made longer because of time loss during an increased number of gear shifts and because of a drop in acceleration performance in the highest gear. For these reasons, compared with the conventional models, the power train of the present invention boosts starting acceleration performance.

As apparent from Figure 8, which shows the relationships between fuel economy and acceleration time that are shown relative to overall reduction ratio in Figures 6 and 7, respectively, in the range of overall reduction ratios between 2.1 and 2.8, either driving feeling or fuel economy improves with changes in the overall reduction ratio. However, when the overall reduction ratio becomes larger than 2.8 or when becoming smaller than 2.1, both driving feeling and fuel economy tend to worsen. Accordingly, overall reduction ratios between 2.1 and 2.8 are effective for both fuel economy and starting acceleration performance.

Referring to Figure 11, which shows the relationship between engine speed and noise, it is revealed that noise increases with a rise in engine speed. Accordingly, the decreased overall reduction ratio and the lowered maximum engine speed for a maximum horsepower makes normal engine noise accordingly smaller, accompanying with enhanced engine quietness. Furthermore, the lowered maximum engine speed, and hence the lowered maximum speed of the supercharger improves the reliability of both the engine 1 and the supercharger 20. It is desirable, in terms of fuel economy and engine output, for the engine 1 to have a displacement between around 2,000 and 2,500 cc. When the engine 1 having even this level of displacement is supercharged, it is powerful comparably to non-supercharged engines whose displacement is of around 3,000 cc. Incidentally, one example of the preferred specifications of the power train of the present invention is shown below.

Transmission (automatic transmission)	
Gear ratio in first gear	2.785

(continued)

Transmission (automatic transmission)	
Gear ratio in second gear	1.545
Gear ratio in third gear	1.000
Gear ratio in fourth gear	0.694
Final reduction gear ratio	3.805
Engine displacement	2,254 cc
Cylinder bore x stroke	80.3 mm x 74.2 mm
Maximum engine output	220 ps/5,550 rpm
Maximum output torque	30 kgf·m/3,500 rpm

In addition, the generally known equations relating to dynamic characteristics are shown below.

Axle output: He

$$He = \{(Pe \cdot Vn \cdot N)/4.5 \times 10^5)\} \cdot e$$

$$= Te \cdot N/716.2$$

Pe: Average effective pressure
Vn: Total engine displacement
N: Engine speed in rpm.
e: Engine cycle coefficient (which is, for instance, 0.5 for 4 cycle engines)

Axle torque: Te

$$Te = \{(Pe \cdot Vn)/(200 \cdot \pi)\} \cdot e$$

$$= 716.2 \cdot He/N$$

Driving force: F

$$F = Te \cdot i_j \cdot FGR \cdot \eta_j / r$$

i_j: Transmission gear ratio (j = 1, 2, 3)
FGR: Terminal reduction ratio
η_j : Driving force transfer efficiency (j = 1, 2, 3,...)
r: Effective tire radius

As indicated by the above equations, the driving force F is related to the axle torque Te, transmission gear ratio i_j, terminal reduction ratio FGR, and driving force transfer efficiency η_j, and effective tire radius r besides, and, in order to attain a certain fixed driving force, if the effective tire radius r is small, the axle torque and/or the overall reduction ratio should be made smaller, and if it is large, they should be made larger. However, effective radii of tires used on various passenger cars fall within the range of around 0.26 to 0.32 m, as shown in the following Table. As long as the power train specifications are defined as described above with taking the effective tire radius to be within the range of 0.26 to 0.32 m, the necessary driving force is attained, certainly providing driving performance.

TABLE

	Passenger car tire size			Tire Radius R (cm)	Effective Radius r = 0.96·R
	Width (cm)	Aspect ratio (H/W in section)	Rim diameter (in)		
Max.	215	65	15	33.0	about 32
Min.	155	65	13	26.6	about 26

As described above, the power train of the present invention is adapted to have an overall reduction ratio in the range of 2.1 to 2.8, smaller than those of the conventional models, for the highest transmission gear so as to provide a maximum engine speed for maximum torque reduced to less than 6,000 rpm., while enabling the supercharged engine to put out boosted torque due to a retardation of closing the intake valves larger than 65 degrees after bottom dead center. The power train thus structured provides a secured output and improved driving performance including starting acceleration while greatly improving fuel economy, enhancing the reliability of both engine and supercharger, and further reducing noises.

It is to be understood that whereas the present invention has been described in detail with respect to a preferred embodiment thereof, nevertheless, various other embodiments and variants may occur to those skilled in the art which are within the scope of the invention as defined by the appended claims. Such other embodiments and variants are intended to be covered by the following claims.

Claims

1. Power train including an engine, which is equipped with a mechanical supercharger and an intercooler disposed in an intake line, and a power transfer line including a transmission operationally connected to said engine, said engine having an intake valve closing time retarded more than 65 degrees of a crank angle after bottom dead centre of an intake stroke,
characterized in that

    said engine provides a maximum output according to a maximum volume of supercharge of said mechanical supercharger,
    said engine has a maximum speed for said maximum output of less than 6,000 rpm, and
    said power transfer line has an overall reduction ratio between 2.1 and 2.8 for the highest gear of said transmission.

2. Power train according to claim 1,
characterized in that

    said mechanical supercharger being of an internal compression type and
    in that said engine having a geometrical compression ratio of greater than 8.5.

3. Power train as defined in claim 2,
characterized in that
said engine having an engine displacement of less than approximately 2,500 cc.

4. Power train as defined in claim 3,
characterized in that
said engine being provided with a plurality of intake ports on each cylinder.

5. Power train as defined in claim 4,
characterized in that
said engine being further provided with a plurality of exhaust ports on each cylinder, with a total opening area of said intake ports being larger than that of said exhaust ports.

6. Power train as defined in claim 5,

characterized in that
said engine being of a double overhead camshaft type.

7. Power train as defined in claim 6,
characterized in that
said transmission having a gear reduction ratio of less than 1.0 for the highest gear.

8. Power train according to any of the preceding claims,
characterized in that
said transmission has four forward gears and in that the corresponding gear reduction ratios have a proportional relationship of approximately 4:2:1,4:1.

Patentansprüche

1. Antriebsstrang, welcher einen Motor, welcher mit einem mechanischen Turbolader und einem in einer Einlaßleitung angeordneten Ladeluftkühler ausgerüstet ist, und einen Kraftübertragungsstrang aufweist, welcher ein mit dem Motor funktionsmäßig verbundenes Getriebe aufweist,
wobei der Motor eine Einlaßventil-Schließzeit aufweist, welche mehr als 65° eines Kurbelwinkels nach dem unteren Totpunkt eines Einlaßtakts verzögert ist,
dadurch **gekennzeichnet**,

daß der Motor eine maximale Ausgangsleistung gemäß eines maximalen Turboladevolumens des mechanischen Turboladers liefert,
daß der Motor eine maximale Drehzahl bei maximaler Ausgangsleistung von weniger als 6000 U/m aufweist, und
daß der Kraftübertragungsstrang ein Gesamtreduktionsverhältnis zwischen 2,1 und 2,8 für den höchsten Gang des Getriebes aufweist.

2. Antriebsstrang nach Anspruch 1,
dadurch **gekennzeichnet**,

daß der mechanische Turbolader vom inneren Kompressionstyp ist, und
daß der Motor ein geometrisches Kompressionsverhältnis größer als 8,5 aufweist.

3. Antriebsstrang nach Anspruch 2,
dadurch **gekennzeichnet**,
daß der Motor einen Motorhubraum von weniger als etwa 2500 ccm aufweist.

4. Antriebsstrang nach Anspruch 3,
dadurch **gekennzeichnet**,
daß am Motor an jedem Zylinder eine Mehrzahl von Einlaßöffnungen vorgesehen ist.

5. Antriebsstrang nach Anspruch 4,
dadurch **gekennzeichnet**,
daß am Motor an jedem Zylinder ferner eine Mehrzahl von Auslaßöffnungen vorgesehen ist, wobei die gesamte Öffnungsfläche der Einlaßöffnungen größer ist als die der Auslaßöffnungen.

6. Antriebsstrang nach Anspruch 5,
dadurch **gekennzeichnet**,
daß der Motor vom Typ eines Doppelnockenwellenmotors ist.

7. Antriebsstrang nach Anspruch 6,
dadurch **gekennzeichnet**,
daß das Getriebe ein Getriebereduzierungsverhältnis von weniger als 1,0 für den höchsten Gang aufweist.

8. Antriebsstrang nach einem der vorhergehenden Ansprüche,
dadurch **gekennzeichnet**,

daß das Getriebe vier Vorwärtsgänge aufweist, und daß die entsprechenden Getriebereduktionsverhältnisse ein proportionales Verhältnis von 4:2:1,4:1 aufweisen.

Revendications

1. Groupe motopropulseur comprenant un moteur à combustion, qui est équipé d'un compresseur mécanique de suralimentation et d'un dispositif de refroidissement intermédiaire placés dans la ligne d'admission, et une ligne de transmission de puissance comprenant une boîte de vitesses opérationnellement couplée audit moteur à combustion, ledit moteur à combustion ayant un retard de fermeture de soupape d'admission supérieur à un angle de vilebrequin de 65 degrés par rapport au point mort bas de la course d'admission,
caractérisé en ce que :

   ledit moteur fournit une sortie maximale fonction d'un volume maximal de suralimentation dudit compresseur mécanique,
   ledit moteur a une vitesse maximale pour ladite sortie maximale de moins de 6000 tours/mn, et
   ladite ligne de transmission de puissance a un rapport de démultiplication global compris entre 2,1 et 2,8 pour le plus fort rapport de ladite boîte de vitesse.

2. Groupe motopropulseur selon la revendication 1, caractérisé en ce que ledit compresseur mécanique de suralimentation est du type à compression interne et en ce que ledit moteur a un rapport géométrique de compression supérieur à 8,5.

3. Groupe motopropulseur selon la revendication 2, caractérisé en ce que ledit moteur a une cylindrée inférieure à 2500 cm^3 environ.

4. Groupe motopropulseur selon la revendication 3, caractérisé en ce que ledit moteur comporte une pluralité d'orifices d'admission sur chaque cylindre.

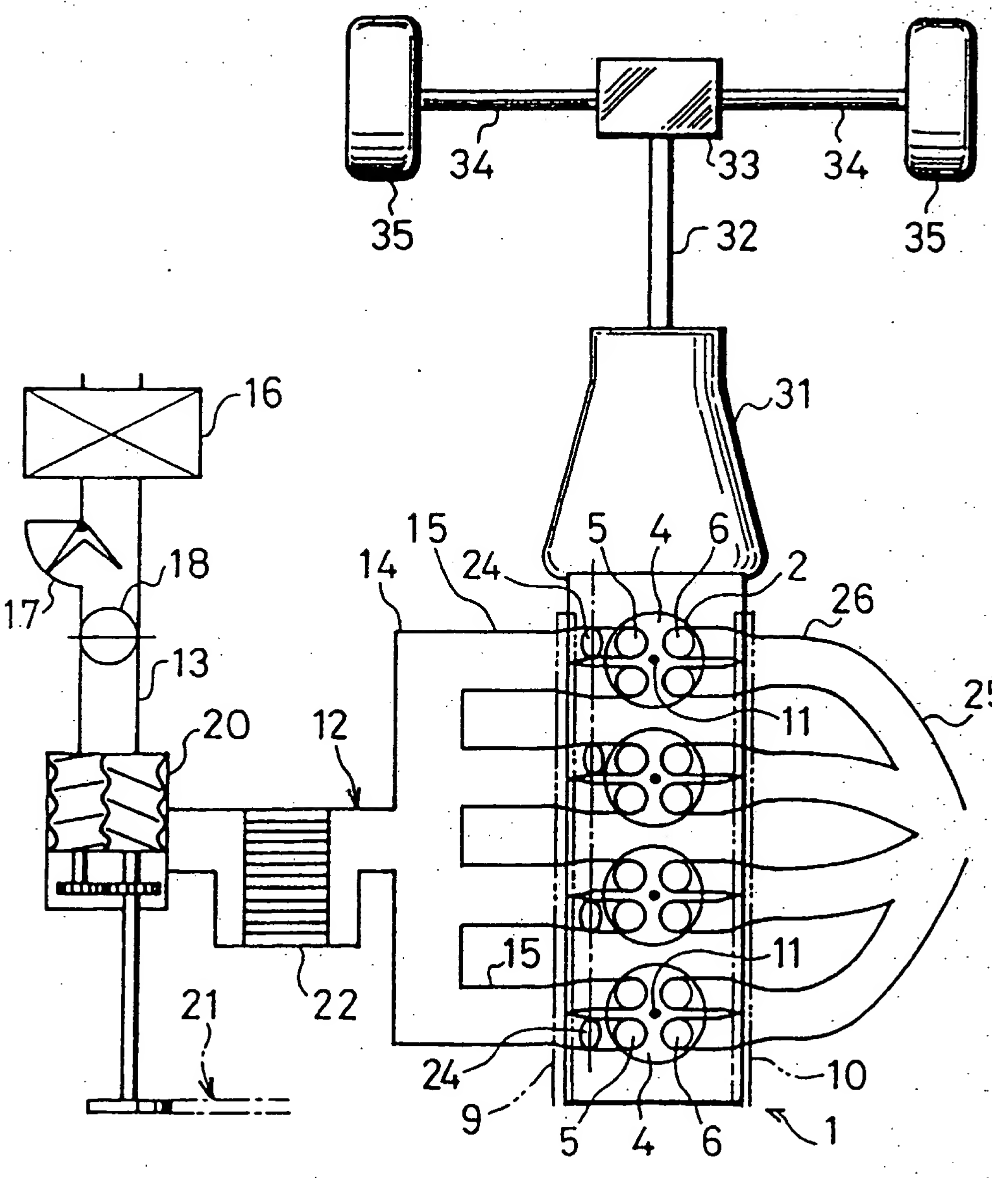
5. Groupe motopropulseur selon la revendication 4, caractérisé en ce que ledit moteur comporte en outre une pluralité d'orifices d'échappement sur chaque cylindre, la superficie totale d'ouverture desdits orifices d'admission étant supérieure à celle desdits orifices d'échappement.

6. Groupe motopropulseur selon la revendication 5, caractérisé en ce que ledit moteur est du type à double arbre à cames en tête.

7. Groupe motopropulseur selon la revendication 6, caractérisé en ce que ladite boîte de vitesses a un rapport de démultiplication inférieur à 1,0 pour le plus fort rapport.

8. Groupe motopropulseur selon l'une quelconque des précédentes revendications, caractérisé en ce que ladite boîte de vitesses a quatre rapports en marche avant et en ce que les rapports de démultiplication correspondants sont liés par une relation de proportionnalité d'approximativement 4:2:1,4:1.

FIG.1



16
17
18
13
20
12
22
21
14
15
24
5
4
6
2
26
25
11
15
11
24
9
5
4
6
10
1
31
32
33
34
34
35
35

FIG. 2

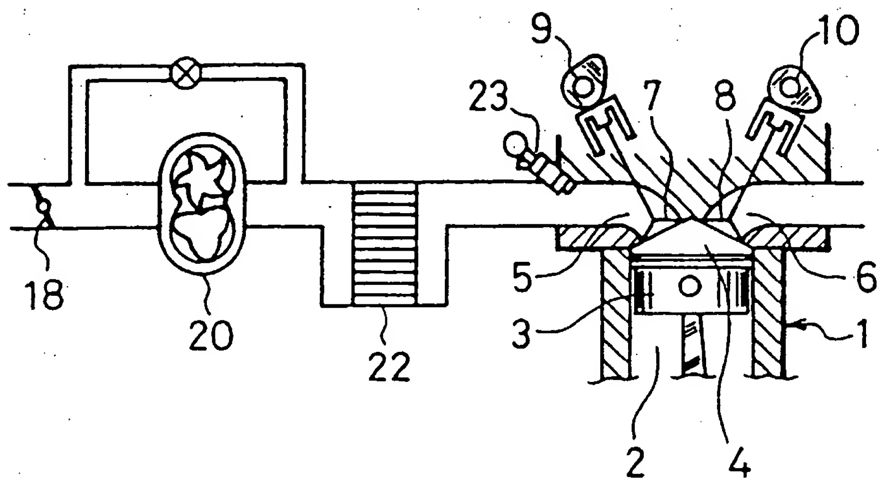


FIG. 3

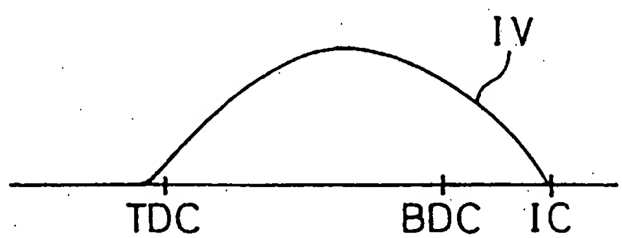


FIG. 4

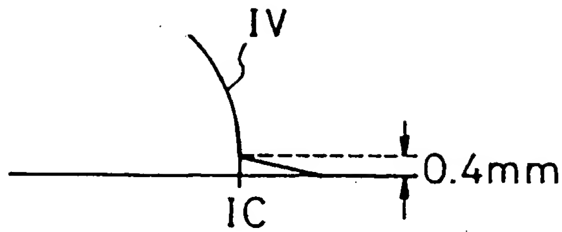

IV
0.4mm
IC

FIG. 5

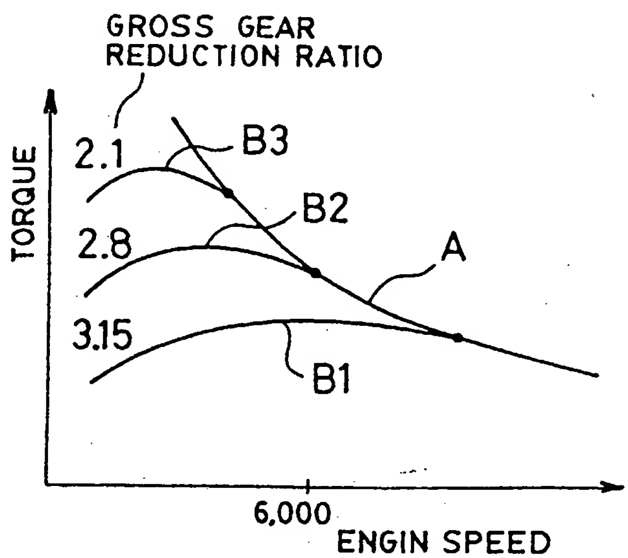

GROSS GEAR
REDUCTION RATIO
TORQUE
2.1
2.8
3.15
B3
B2
A
B1
6,000
ENGIN SPEED

FIG. 6

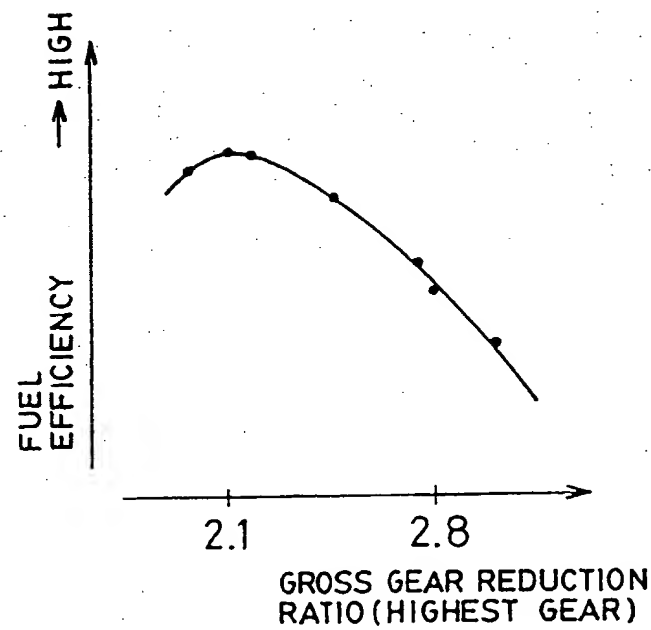

FUEL EFFICIENCY → HIGH
2.1
2.8
GROSS GEAR REDUCTION RATIO (HIGHEST GEAR)

FIG. 7

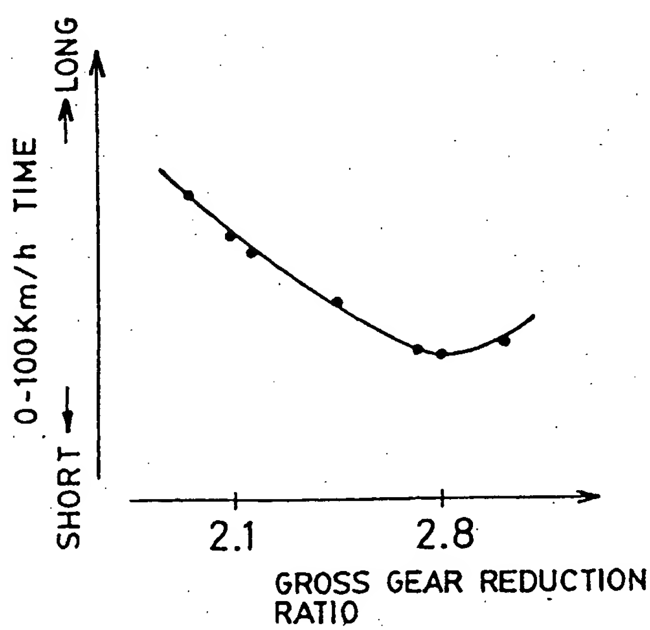

SHORT ← 0-100Km/h TIME → LONG
2.1
2.8
GROSS GEAR REDUCTION RATIO

FIG.8

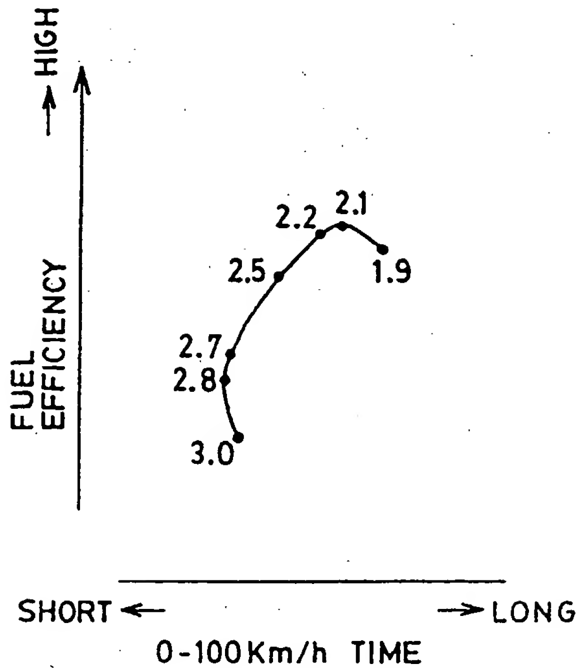

HIGH
FUEL EFFICIENCY
2.1
2.2
2.5
1.9
2.7
2.8
3.0
SHORT
LONG
0-100Km/h TIME

FIG.9

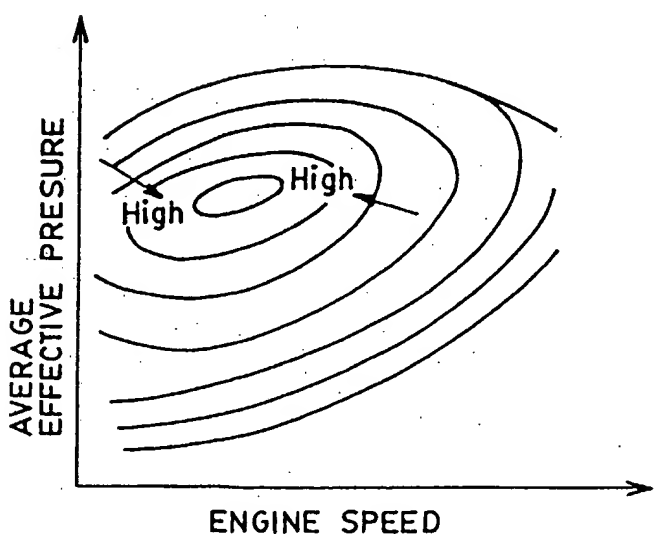

AVERAGE EFFECTIVE PRESURE
High
High
ENGINE SPEED

FIG.10

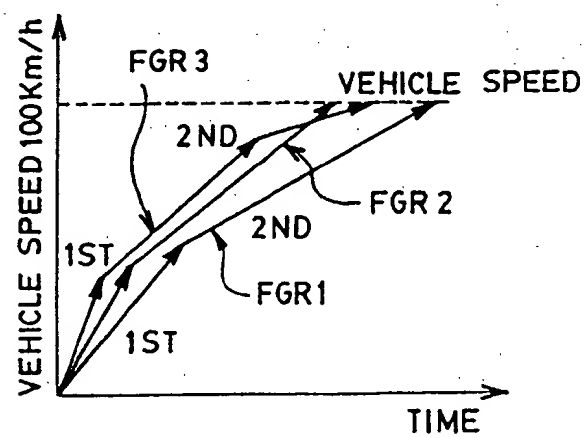
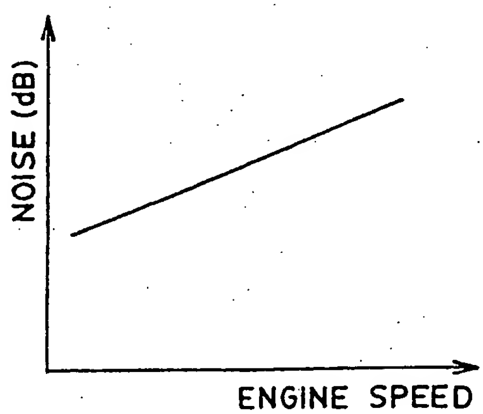

VEHICLE SPEED 100Km/h
FGR 3
VEHICLE SPEED
2ND
FGR 2
2ND
1ST
FGR1
1ST
TIME

FIG.11


NOISE (dB)
ENGINE SPEED